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MASTER

DYNAMIC SURFACE-PRESSURE INSTRUMENTATION FOR RODS IN PARALLEL FLOW

by

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ABSTRACT

Methods employed and experience gained in measuring random fluid boundary layer pressures on the surface of a small diameter cylindrical rod subject to dense, nonhomogeneous, turbulent, parallel flow in a relatively noise-contaminated flow loop are described. Emphasis is placed on identification of instrumentation problems; description of transducer construction, mounting, and waterproofing; and the pretest calibration required to achieve instrumentation capable of reliable data acquisition.

NOMENCLATURE

a_1	Rod amplitude at fundamental frequency f_1
d	Rod diameter
d_t	Transducer diameter
D	Outside diameter of annular region
D_h	Hydraulic diameter
f_1	Fundamental rod frequency
p_s	Stagnation pressure
T	Turbulent eddy time period
V	Mean flow velocity
ρ_f	Fluid mass density

INTRODUCTION

The relatively high velocity, heat removing coolant in reactors and heat exchangers represents a source of turbulent energy which is often capable of exciting structural components which it contacts. In particular, rods subject to nominally parallel flow is a common geometry. The random wall pressure fluctuations beneath the turbulent boundary layer on the rod are believed [1] to be the primary excitation source for the flow geometries of light water reactors in single phase flow. Sufficient structural methodology is available for purposes of analysis; however, reliable and representative fluid forcing functions have not been completely formulated [2]. Even for the simplest configuration of a very long rod in an infinite flow field, with minimal in-stream turbulence, the characterization of the pressure fluctuations beneath the fully developed boundary layer on the rod is not complete [3]. Application of the available characterization in a vibration analysis is questionable, because few reactor components are subject to such ideal flow conditions. Predictions of reactor component response based on the pressure fields of ideal flow conditions may underestimate actual response by two orders of magnitude [8]

Typically in reactor systems the flow is channeled around the rod and/or severe in-stream flow turbulence is created by discontinuities in component geometries and spacing grids. Toward obtaining more realistic assessment of rod vibrations and pressure fields, a rod in an annular flow region has been studied by many investigators. In general, vibration levels are larger than those predictable for ideal flow conditions but still small from the standpoint of stress levels and fatigue. Wear at supports appears to be the major concern [2]. Also, test results [4,5,8] indicate that loop noise and/or upstream flow disturbances can increase vibration levels and wall pressure levels by an order of magnitude.

Based on the form of the governing equations for incompressible flow, the pressure fluctuations at the rod surface are known [3] to be influenced by the fluid momentum changes everywhere in the flow channel: the turbulence in the boundary layer as well as any introduced in the main flow stream. However, the relative influence of each source is not available either from solutions to the equations of motion or from experimental data. Thus, as a complement to previous forcing function characterizations of relatively rigid rods in an annular region subject to fully developed turbulent flow [4,9], a series of measurements was planned to characterize the surface pressures of a relatively flexible central rod subject to annular water flow with several defined upstream disturbances. In creating the upstream disturbances nominally parallel flow conditions were maintained, but significantly greater turbulence was introduced than occurs naturally in fully developed pipe flow.

The instrumentation and methods developed to make the pressure measurements are presented here along with the resolution of several instrumentation problems created by the test conditions. Because the choice of instrumentation and methods employed is primarily dictated by the test configurations and requirements, a brief description of both are provided.

TEST DEFINITION

A rod was located in a test section at the end of a long pipe conveying fully developed turbulent water flow. The test section and rod form an annular region, Fig. 1. The 1.22 m long rod was fabricated from 1/4 inch schedule 80 pipe stock with outside and inside diameters of 13.7 and 7.67 mm, respectively. The downstream (upper) end of the rod was fastened to streamlined support fins which centrally locate the rod in the pipe and provide a fixed-end condition. The upstream (lower) end of the rod was seated in an O-ring mounted in streamlined support fins which resulted in a simple supported end condition. The rod size was not chosen to represent any particular reactor component, although it is typical of fuel rods in several reactors.

A V-shaped stainless steel channel serves as a pressure boundary for the annular region, see Fig. 1, and was designed to obtain a flexurally-rigid test section. Static test section pressures of 207 KPa were typical. The test section was separated from the rigid loop piping by rubber vibration isolators and mounted to a relatively massive test stand. Although the test model was structurally isolated from external disturbances, the location of the 3.15×10^{-2} m^3/s constant speed water pump had to be changed, as will be discussed later, to reduce noise transmission through the fluid and structure.

The theory for the random response of rods in parallel flow [4] makes clear that characterization of the pressure field forcing function requires knowledge of the cross power spectrum between every axial and circumferential location, especially in the range of the structural vibration frequencies. Since pressure measurements are difficult and practically cannot be made between every spatial location and frequency, usually analytical models of the pressure field are assumed which require a minimum number of pressure measurements for characterization. Most often a spatially homogeneous pressure field is assumed and measurements are made at relatively high frequencies over a single surface area of the rod for which the pressure field shows significant correlation [3]. Normal wide band correlation lengths are only on the order of boundary layer displacement thickness, δ^*, which would be on the order of 0.64 mm if ideal annular flow [7] occurred in the test section.

For the case under study, the pressure field was expected to be homogeneous around the circumference of the rod but not along the rod axis because of the decaying entrance disturbances. The frequency range of interest was below 300 Hz, a range covering the lowest rod vibration frequencies. This frequency range is often neglected in many studies of boundary layer pressure fluctuations because of the presence of large amplitude spurious loop noises. The theory for random response of rods also makes clear that if significant levels of vibrations are to occur, then the pressure field correlation lengths will have to be on the order of the vibration wavelength, the rod length. Small boundary layer correlation lengths on the order of the displacement thickness are very ineffective in producing rod motion. As a first step in characterization of the pressure field, the four axial locations shown in Fig. 1 were chosen to assess the axial homogeneity of the pressure field and the existence of wide and/or narrow frequency band correlation on the order of the fundamental vibration mode wavelength. If such correlation lengths exist they probably are generated by upstream disturbances or gross unsteadiness in the flow rather than by the rod boundary layer.

ANTICIPATED PROBLEMS

Based on past experience a number of problems in making reliable measurements could be foreseen, the most serious of which will be considered first. Plane sound waves originating at the pump, upstream valves, discontinuities in piping, and vibration of piping components are referred to as far field noise. The far field noise propagates through the test section and results in contamination of the wall pressures in the low-frequency range of interest. Because of the commercial pumping system employed, the elimination of the far field noise, which could be further aggravated by the use of turbulence producing grids, was a major concern. The previously successfully employed method [5,9] of using diametrally opposed pressure transducers to eliminate the effects of far field noise was selected.

To understand the technique consider a transverse cross section of the rod shown in Fig. 2. For structural analysis of beams the resultant force on the rod per unit length is most important. For example, the force in the vertical direction F_v is the integrated effect of the pressure field around the circumference.

$$F_v = \int_0^{2\pi} p(x,\theta,t)\cos\theta \; r_i d\theta = \int_0^{\pi} (p_A - p_B)\cos\theta \; r_i d\theta \qquad (1)$$

where p_A and p_B are the pressures at diametrally opposed positions around the rod circumference. Assuming that the far field noise propagates through the test section as a one-dimensional plane wave, the far field contribution to the pressure is correlated in both amplitude and phase around the circumference of the rod and is canceled out in taking the differences in Eq. (1). To assure that the noise contaminated signals could be subtracted to recover the excitation pressures, expected to be up to one hundred times smaller, rather elaborate calibration procedures were employed and will be discussed after the transducers employed are described.

Subtraction of diametrically opposed pressure transducer signals, although beneficial in reducing far field noise contamination, can be undesirable in that transducer sensitivity to rod strain tends to be doubled because equal and opposite strains occur on opposite sides of the rod. The "apparent pressures" due to rod strain are usually confined to narrow bands around the rod natural frequency and can be filtered out if the strain effects are maintained on the order of the pressure effects, a more difficult task when flexible rods are employed because of the greater strain response. In the discussion of the transducer construction and calibration both a successful and unsuccessful mounting attempt will be illustrated.

Attaining a degree of flushness and continuity of the transducer with the test rod surface to minimize disturbance of the flow is unavoidably linked with transducer frequency resolution and isolation from rod strain. Crudely, a turbulent eddy with time period T, or frequency 1/T, being convected at the mean velocity V will have its effects canceled out on a transducer of diameter d_t unless $VT > d_t$. In actuality, this condition only serves as an upper bound on transducer size since the convection velocity of the turbulent eddies is usually less than the mean velocity. The best resolution has been attained by combinations of subsurface microphones connected to the flow field via pinholes (the order of 1 mm diameter); however, the effects of such a surface discontinuity on the flow field have not been completely resolved [3]. Pressure oscillations due to flow over cavities can be limited to frequency ranges beyond those of interest by controlling the size and shape of the pinhole cavity, but the possibility of local surface roughness effecting the pressure field [6] cannot be ruled out.

Flush mounting the transducer in the rod surface permits reduction in surface roughness, but some high frequency resolution is lost because the sensing surfaces of the transducer in contact with the flow is inherently larger. Also, the reduction in surface roughness cannot be complete since some degree of separation of the transducer from the rod must exist to avoid extreme transducer sensitivity to rod strain and/or vibration. A compromise must be reached based on the particular requirements of each application.

Flush mounted transducers were employed primarily because they allow maintaining surface continuity to a reasonable degree. However, they also provide a practical means of instrumenting diametrally opposite sides of the small test rod described previously. The overall size of the transducers, $\sim$3 mm in diameter, was expected to be able to resolve pressure fluctuations in the structural frequency range of interest, below 300 Hz, at the test flow velocity V of 1.5 to 10 m/s. Surface mounting of the transducers produced discontinuities up to 0.125 mm but such roughnesses would not appear to effect the flow field in the frequency range of interest at correlation lengths on the order of the rod length. Measurements of very local roughness effects for ideal boundary layer flows [6] show the correlation lengths and power spectrum magnitudes are effected only when the surface roughness is of the order of the correlation length and the displacement boundary layer thickness, respectively. Even if a displacement layer is meaningful in the highly disturbed flow under consideration, it would be 5 to 10 times larger than the surface roughness present.

The order of magnitude of the spurious pressures which will be created by rod vibration can be estimated knowing the rod natural frequencies f_n and amplitudes a_n. Considering the rod of Fig. 2 to be vibrating transversely with amplitude a_1 at the fundamental natural frequency f_1, the maximum relative velocity between the rod and a still fluid will be $2\pi f_1 a_1$, with an associated stagnation pressure p_s of $1/2\ \rho_f(2\pi f_1 a_1)^2$ where ρ_f is the fluid mass density. For typical test condition of f_1 = 30 Hz and a_1 = 0.10 mm in water the stagnation pressure computed in this manner is p_s = 0.183 Pa. For an ideal boundary layer [3], the wide band root mean square pressure is on the order of 0.01 times the dynamic pressure and the main contribution will occur primarily below a maximum frequency of $f_{max} = \frac{1}{2\pi} V/\delta*$. For V = 3 m/s and $\delta*$ = 0.64 mm the narrow band pressures expected will be on the order of 0.1 Pa/Hz, assuming pressures are distributed uniformly over the frequency range. This is the same order as the rod vibration induced pressure, thus the narrow bands around the rod natural frequencies probably will have to be removed, filtered, from the pressure field data.

Vibration of the test section can introduce even greater spurious pressure fluctuations than that associated with vibrations of the rod. Test section vibrations accelerate the radial column of water between the pressure transducer surface and the outside boundary of the annular region, Fig. 2, nominally at the acceleration of the test section, a_t. For this case, the pressure exerted by the transducer to accelerate the water column of length (r_o-r_i) is $p_t = \rho_f(r_o-r_i)a_t$. For the largest (r_o-r_i) under consideration, approximately 19 mm, an a_t which produces spurious pressures on the order of the total root mean square pressure at 3 m/s would be a_t = 0.37 g's. Maintenance of such low vibration levels would require an almost impossible isolation task. For the test loop, acceleration levels of 0.2 g's rms were measured for loop operating parameters producing the worst vibrations. As for rod vibrations, the test section natural frequencies will have to be identified and narrow bands filtered from the pressure field data.

When pressurized water is the test fluid and piezoelectric pressure transducers are employed, then waterproofing the transducers and connections becomes a major concern because very high resistance to ground, on the order of hundreds of megohms, must be maintained. For such cases, waterproofing amounts to the creation of multiple barriers that the water molecules must circumvent to establish a ground path. By providing enough barriers a reasonable time for testing can be obtained, conceding that intolerable grounding will eventually occur and re-waterproofing will be required. The testing time and number of barriers obtainable depends a great deal on the skills and techniques available for fabricating and mounting the transducers.

TRANSDUCERS

The requirements for the pressure measuring transducers were extremely rigid. In addition to resolving wide band dynamic pressures on the order of 69 Pa rms, they needed to have a small pressure sensing area of ∿3 mm diameter flush with the rod surface, to be waterproof to 207 KPa, and two transducers had to fit diametrically-opposite of one another in a 1/4 inch schedule 80 pipe. It was also planned to have four pairs of these transducers in one rod to characterize the nearfield pressures over most of its length. Transducers which would fill these requirements could be of either the piezoelectric type or of piezoresistive semiconductor material configured in a full Wheatstone bridge.

A survey of commercially available transducers found several which met some of the required specifications, but all would require modifications before they could possibly be used for the test. Since a minimum of 8 transducers was required and the usefulness of modified commercially available transducers was in question, the decision was made to custom fabricate piezoelectric crystal transducers in place flush with the test rod surface. This decision was promoted because of past success with this approach [9]. A schematic of the miniature transducer assembly is shown in Fig. 3.

Adequate waterproofing of the custom built pressure transducers was critical to maintain high resistance (megohms) to ground. Thermal setting potting compounds were discarded because of the low Curie temperature of the crystals used for the pressure transducers. Most RTV silicon compounds waterproof adequately, but the quality of their adhesion to stainless steel is poor. After testing a number of potting compounds and adhesives, an adequate waterproofing technique was developed.

After the transducer was mounted in the well on the rod surface, Fig. 3, the well was lined with Caulk Grip dental cement which adhered well to the stainless steel and had good waterproofing characteristics. However, the dental cement is very rigid and if used alone would couple the transducer and rod strain. The transducer was potted in a two-part fuel tank sealant compound (PR-385 manufactured by Products Research and Chemical Corporation) which was very flexible and adhered to the dental cement. This potting was contoured to the curvature of the test rod and allowed to cure. Finally, the surface of the potted pressure transducer was treated with a micro-crystalline wax having zero water absorption characteristics. Test leads were routed out the top of the test rod through flexible tubing secured and made leak tight at the rod top. This tubing was then brought through a compression type fitting at the test chamber outlet area so as not to effect downstream flow conditions.

Because of problems which could arise with transducer fabrication in place, such as high base strain sensitivity, poor low frequency response, or low charge sensitivity; parallel efforts were made to modify a commercially available pressure transducer. An Entran model EPA S125E-30SW was chosen including custom manufacturer modifications of sealing the transducer by electron beam welding the pressure sensitive diaphragms to transducer housings and shortening the transducers to a total length of 5.33 mm. The mounting of the commercial transducer is illustrated in Fig. 4. The mounting holes in the rod were reamed slightly larger at the top to insure that the sensitive diaphragm of the pressure transducer was not coupled to straining of the rod surface. Only the less strain sensitive base of the transducer was glued near the inside diameter of the rod. Access holes were drilled to aid in transducer placement and for final waterproofing at the rear of the transducers. After all eight transducers were mounted and waterproofed these access holes were filled and faired to conform to the test element surface.

CALIBRATION

Calibration of the custom built piezoelectric pressure transducers required the construction of a source for generating small sinusoidal pressure fluctuations in the presence of much larger static pressures, because the charge response of the piezoelectric crystals are known to vary over relatively large pressure changes. A diaphragm and actuator were attached to a pressure chamber which could be fit around the rod at the different pressure transducer locations, as shown in Fig. 5. The actuator was spring loaded to balance the static pressure and was driven by an electrodynamic shaker. Calibration of the rod transducers was to be made by comparison to a standardized piezoelectric pressure transducer mounted in the side of the pressure chamber. The standard transducer's output was employed as feedback to a Spectral Dynamics Servo System to maintain a constant dynamic pressure amplitude, selected between 69 - 690 Pa, as the frequency was swept from 10 to 300 Hz.

The sensitivities of the pressure transducers were found to be large, on the order of 1 picocoulomb per 69 Pa at a static pressure of 138 KPa. Changes of 69 KPa in static pressure changed the calibration sensitivity approximately 10%. For a given static pressure the sensitivity remained nearly constant over the frequency range, Fig. 6, except large spurious peaks were present below seventy Hertz due to rod strain-transducer coupling. The straining effect became more apparent when the outputs of the two opposite transducers were subtracted as they would be during testing to eliminate far field noise. The effects were too severe to allow recovery of the pressure signal and attention was focused on determining whether the careful mounting of the piezoresistive pressure transducers eliminated similar problems.

Because the output of a piezoresistive transducer is linear, even with large changes in pressures, a static calibration is sufficient. Their pressure voltage sensitivity agreed with the factory supplied calibration to within 1 to 1.5%, and their average sensitivity was 0.326 mv/KPa.

The existence of "apparent pressures" due to rod strain was investigated next. The rod with the transducers mounted was bent so as to put each transducer in compression or tension. The transducers near the ends of the test element, or at the point of least strain, indicated an induced apparent pressure of from 0.8 to 3.3 KPa per mm of rod deflection. Transducers situated at the maximum strain points indicated pressures of 5.5 to 11.1 KPa per mm of rod deflection. Because rod deflections were expected to be very low, magnitudes of 0.05 mm or less, these apparent pressure levels were acceptable.

The final step was to verify that the calibration and amplification of the pressure transducer signal was accurate enough to eliminate far field noise (plane sound waves) by subtraction of opposing pressure signals. Figure 7 shows the test setup. The entire test element was submerged in water and subjected to a far field acoustic pressure field generated by a speaker. The transducer outputs were compared in real time and the frequency domain as shown in Fig. 8. After signal conditioning with an Unholtz-Dickie Model D22MPB Bridge Conditioner and Amplifier, the differences of the signals and the rms pressure levels were determined using a Hewlett-Packard 5451/71B Fast Fourier Analyzer. For a random pressure field of 2.8 KPa and 2.58 KPa rms over a frequency range of 0 to 500 Hz, 95 percent of the far field generated pressure was eliminated.

EXAMPLE TEST RESULTS

While some care was given to structural isolation of the test section, the original facility pump and valve caused considerable noise in the pressure signal, especially in the frequency range of interest. Speculation was that the pump caused loop piping upstream of the test section to vibrate which in turn transmitted sound waves through the water column to the pressure transducers. Some of these effects were eliminated by employing an alternate pump and control valve which were separated from the test section structurally by 20 meters of highly damped, cracked concrete floor and acoustically by a thirty cubic meter accumulator tank in the water line.

Typical noise reductions attained for a single pressure transducer signal p_2 are shown in the power spectrums of Fig. 9. Significant changes in the location and width of several frequency peaks are apparent. Quantitatively, with the original pump the rms pressure level from 0 to 500 Hz was 1.75 KPa, while with the alternate pump and same flow rate and static pressure the rms pressure level was 1.4 KPa. For other operating conditions the differences were both larger and smaller, but in all cases the alternate pump produced a decrease in far field noise at the test section.

Even with separation of the pump and control valve, vibration of the test section occurred. While not large by most standards, it was the source of spurious contributions to the very small pressure signals that were being measured. Because the natural frequencies of the test section could be determined, the remaining spurious contributions were eliminated during data reduction. For example, in determining narrow and wide band rms pressure levels, the identified power from the test section natural frequency peaks was deleted during integration of the power spectrums.

Typical separate power spectrums of opposing pressure transducers 1 and 2 are shown in Fig. 10a and 10b, while Fig. 10c shows the subtracted pressure transducer signals. The separate signals allow easy identification of the presence of the spurious contribution of the test section vibration. The subtracted signals show the very significant reduction in far field noise. Note the doubling of non-far field noise which must be accounted for [9] in data analysis. Quantitatively, the rms pressure levels of the separate pressure signals p_1 and p_2 were 0.74 KPa and 0.781 KPa, respectively, while that of the subtracted pressure signals was 0.29 KPa. The strain induced pressures at the rod natural frequency of 0.048 KPa rms remains to be deleted, but its presence is quite obvious. Spurious contributions due to test section motion were present but negligible in this calculation of the rms pressure, which typically was the case. All data was reduced in a similar manner.

CONCLUSIONS

Obtaining valid dynamic pressures in a commercial water loop requires the resolution of problems in both the areas of instrumentation and data analysis. Inevitably methods of solving such problems must be individualized to each experiment.

Specific hardware was selected to measure surface pressures on small diameter rods in parallel flow. The detailed selection procedure given was based on avoidance of transducer interference with the pressure field being measured and attainment of sufficient transducer amplitude and frequency sensitivity. Also strain coupling between the rod and transducer was shown to be a large potential source of error. A successful mounting technique avoiding strain coupling was presented along with methods of waterproofing.

Evidence was presented that spurious loop far field noise due to valves, motors, pipe bends and other geometric discontinuities are present in most pressure field data and often dominate. A method of isolating the low level pressures associated with the rod turbulent layer from the far field noise was presented. Essentially two diametrically opposed transducers were employed to eliminate the far field noise assumed to be propagating as plane waves. A calibration was performed to show the method worked for a known source of plane wave noise. Finally, the instrumentation and data analysis methods were employed in the determination of rms pressure levels for selected test data.

REFERENCES

1. Paidoussis, M. P., "Vibrations of Cylindrical Structures Induced by Axial Flow," J. Eng. Ind. 96I, 547-553 (1974).

2. Mulcahy, T. M., and Wambsganss, M. W., "Flow-Induced Vibration of Nuclear Reactor Components," The Shock and Vibration Digest 8, 33-45 (1976).

3. Wilmarth, W. W., "Pressure Fluctuations Beneath Turbulent Boundary Layers," Annual Review of Fluid Mechanics 7, 13-38 (1975).

4. Chen, S. S., and Wambsganss, M. W., "Parallel-Flow-Induced Vibration of Fuel Rods," Nucl. Eng. Design 18, 253-278 (1972).

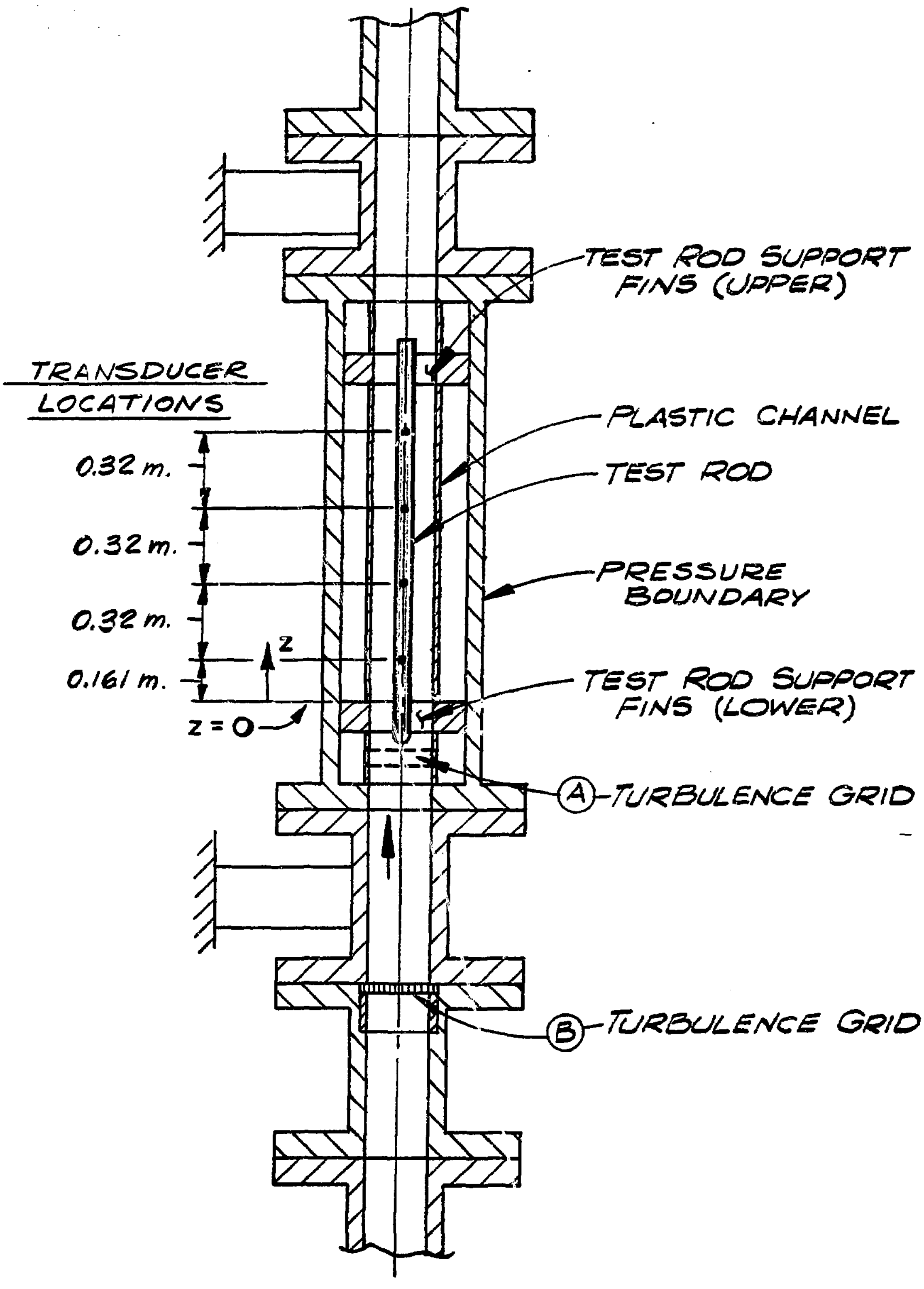
5. Ohlmer, E., Russo, S., and Schwemml, R., "Investigation of an Analytical Model for Parallel Flow Induced Rod Vibrations," Nuclear Eng. Design 22, 272-289 (1972).

6. Aupperle, F. A., and Lambert, R. F., "Effects of Surface Roughness on Measured Wall-Pressure Fluctuations Beneath a Turbulent Boundary Layer," J. Acoust. Soc. Am. 47(1), 359-370 (1970).

7. Brighton, J. A., and Jones, J. B., "Full Developed Turbulent Flow in Annuli," J. Basic Eng., Trans. ASME 86, 835-844 (1964).

8. Reavis, J. R., "Vibration Correlation for Maximum Fuel-Element Displacement in Parallel Turbulent Flow," Nucl. Sci. and Eng. 38, 63-69 (1969).

9. Wambsganss, M. W., and Zaleski, P. L., "Measurement Interpretation and Characterization of Near Field Flow Noise," Proceedings of the Conference on Flow Induced Vibrations in Reactor Components, Argonne National Laboratory Report ANL-7685, 112-137 (1970).



TEST ROD SUPPORT FINS (UPPER)
TRANSDUCER LOCATIONS
PLASTIC CHANNEL
TEST ROD
0.32 m.
0.32 m.
PRESSURE BOUNDARY
0.32 m.
Z
0.161 m.
TEST ROD SUPPORT FINS (LOWER)
Z = 0
A-TURBULENCE GRID
B-TURBULENCE GRID

Fig. 1

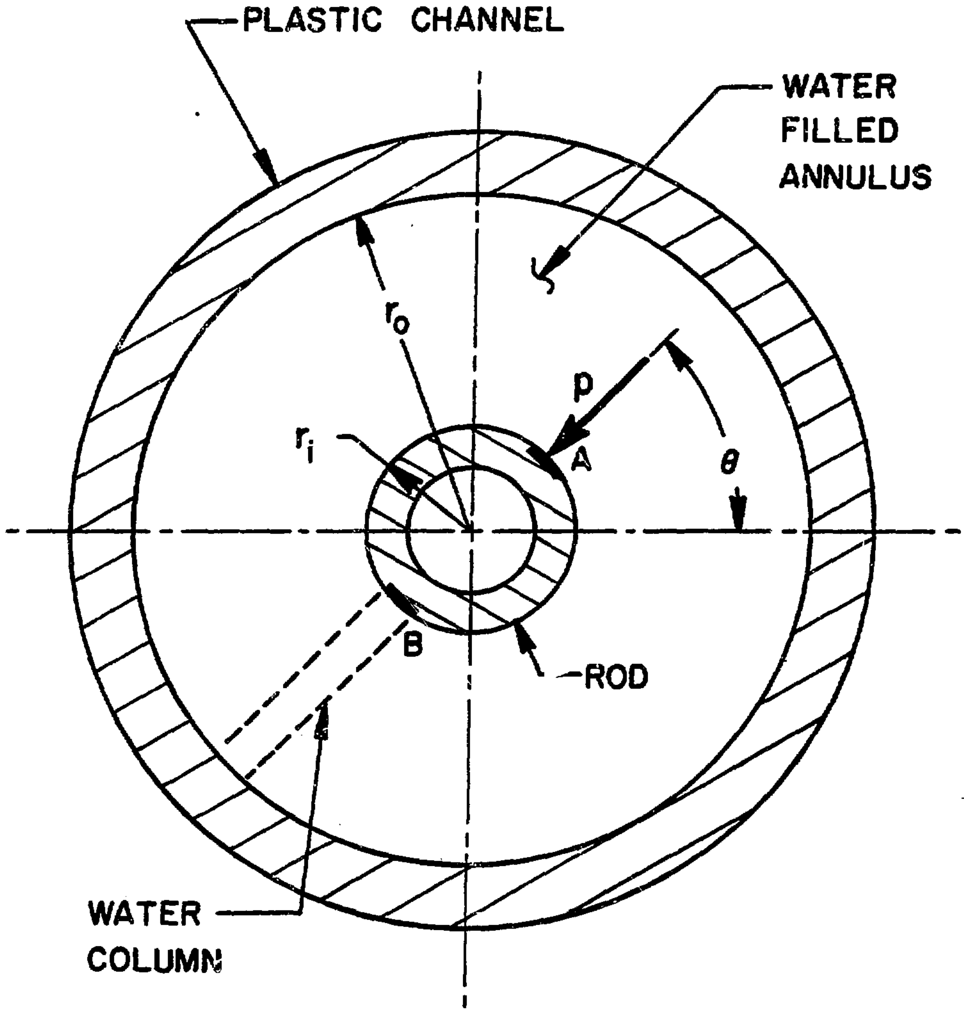

PLASTIC CHANNEL
WATER FILLED ANNULUS
r_o
p
A
θ
r_i
B
ROD
WATER COLUMN

Fig. 2

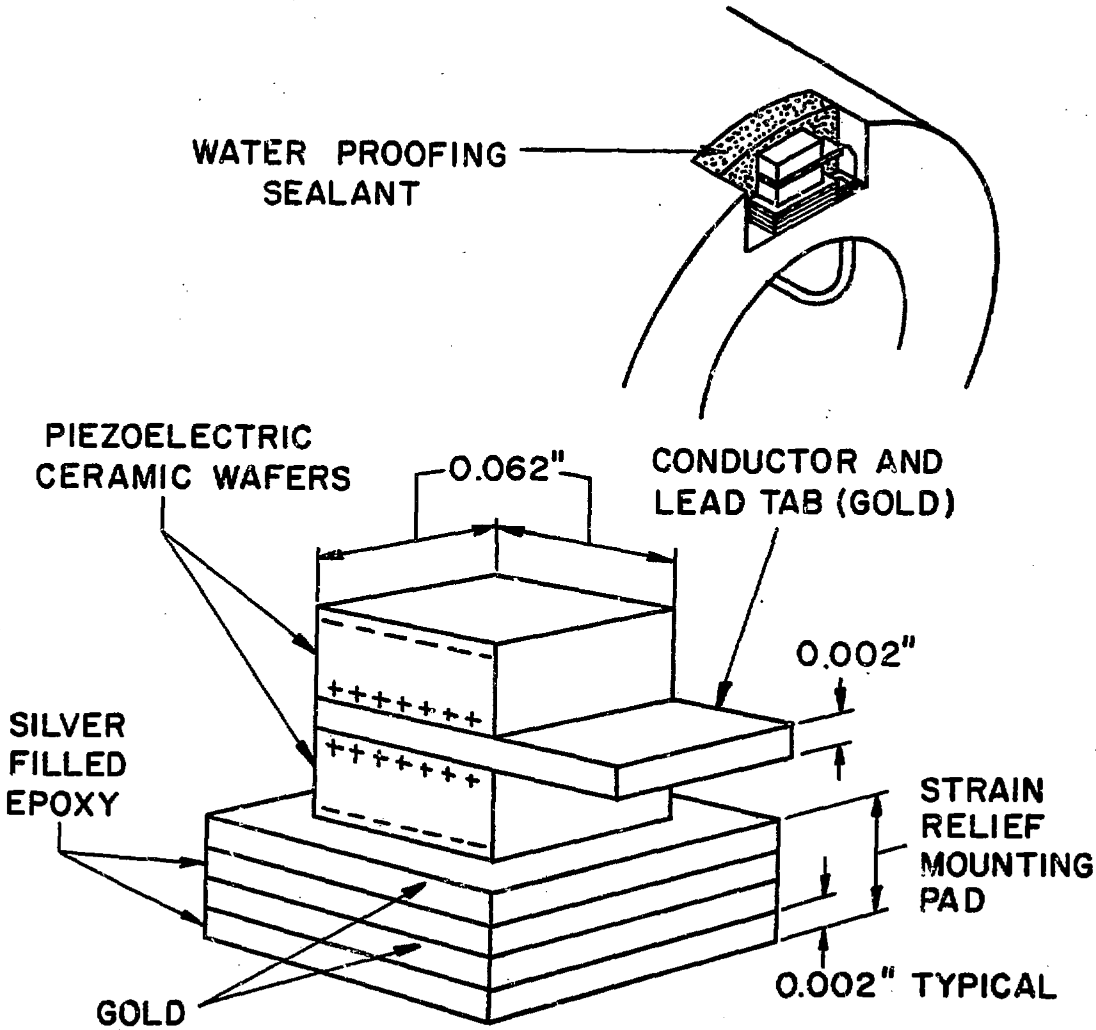

WATER PROOFING SEALANT
PIEZOELECTRIC CERAMIC WAFERS
0.062"
CONDUCTOR AND LEAD TAB (GOLD)
0.002"
SILVER FILLED EPOXY
STRAIN RELIEF MOUNTING PAD
GOLD
0.002" TYPICAL

Fig. 3

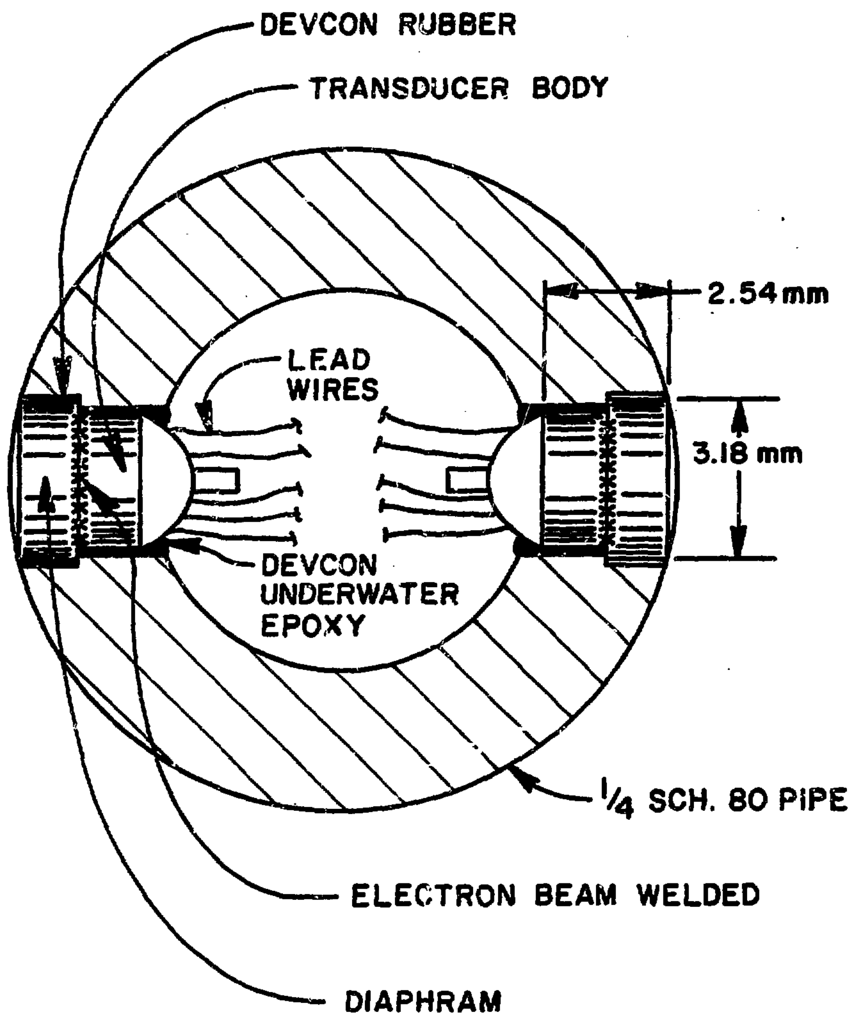

DEVCON RUBBER
TRANSDUCER BODY
2.54 mm
LEAD
WIRES
3.18 mm
DEVCON
UNDERWATER
EPOXY
1/4 SCH. 80 PIPE
ELECTRON BEAM WELDED
DIAPHRAM

Fig. 4

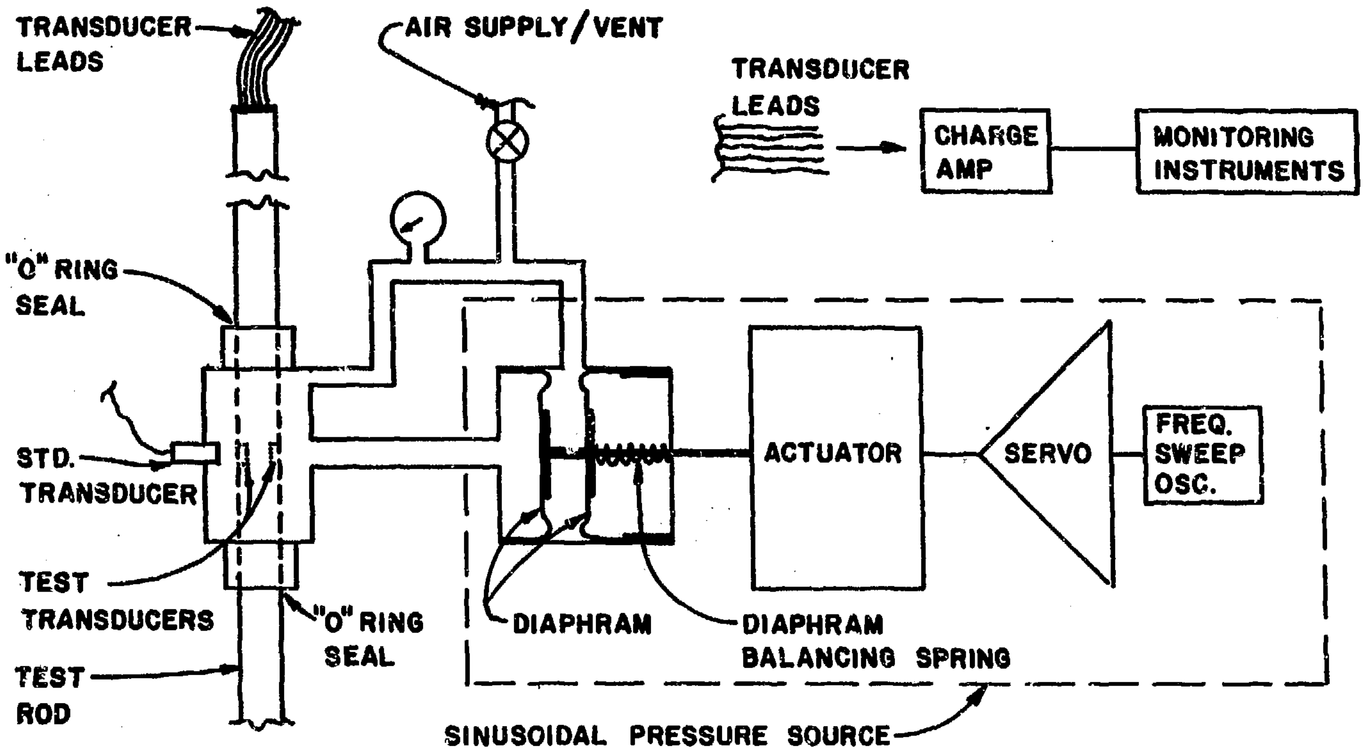

TRANSDUCER LEADS
AIR SUPPLY/VENT
TRANSDUCER LEADS
CHARGE AMP
MONITORING INSTRUMENTS
"O" RING SEAL
STD. TRANSDUCER
ACTUATOR
SERVO
FREQ. SWEEP OSC.
TEST TRANSDUCERS
"O" RING SEAL
DIAPHRAM
DIAPHRAM BALANCING SPRING
TEST ROD
SINUSOIDAL PRESSURE SOURCE

Fig. 5

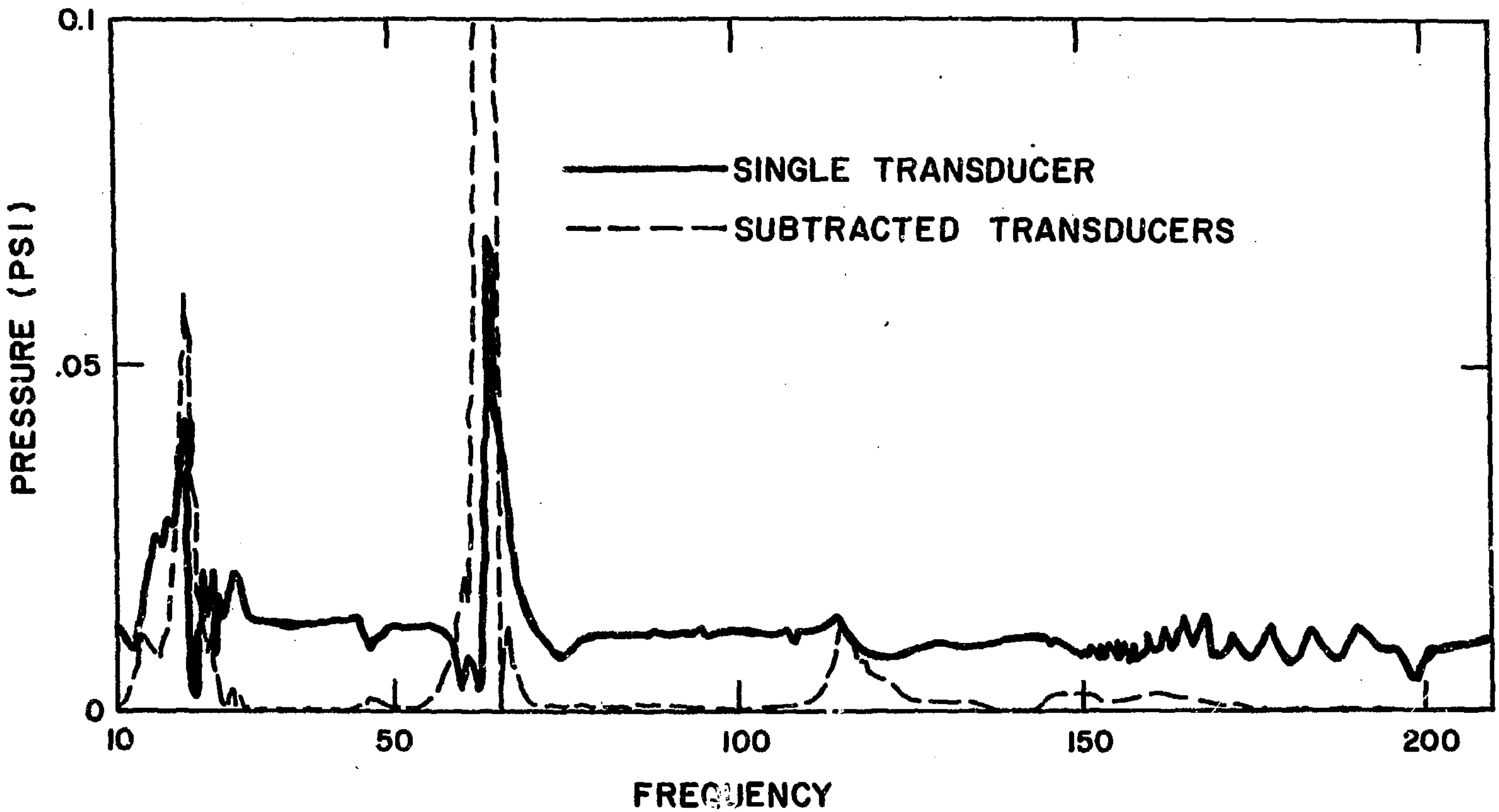

0.1
.05
0
PRESSURE (PSI)
SINGLE TRANSDUCER
SUBTRACTED TRANSDUCERS
10
50
100
150
200
FREQUENCY

Fig. 6

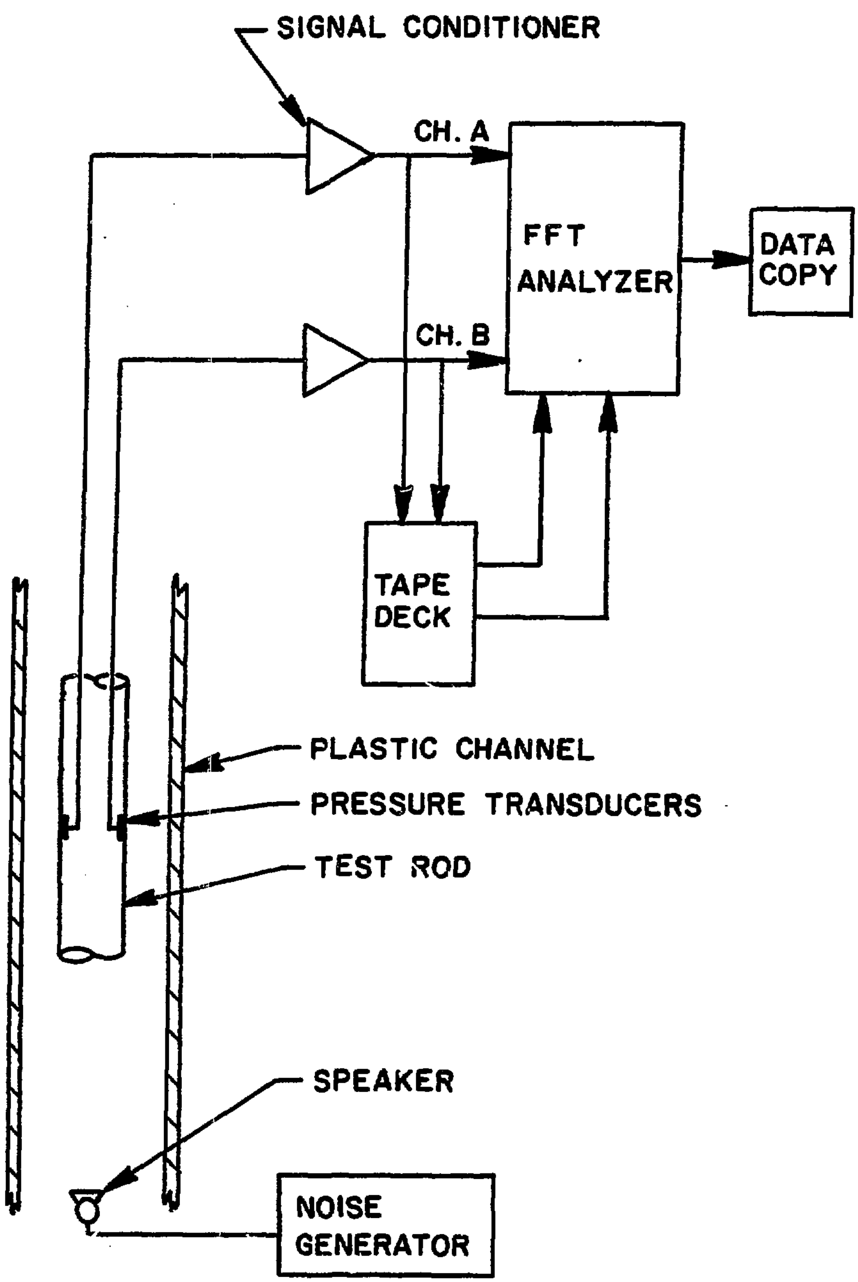
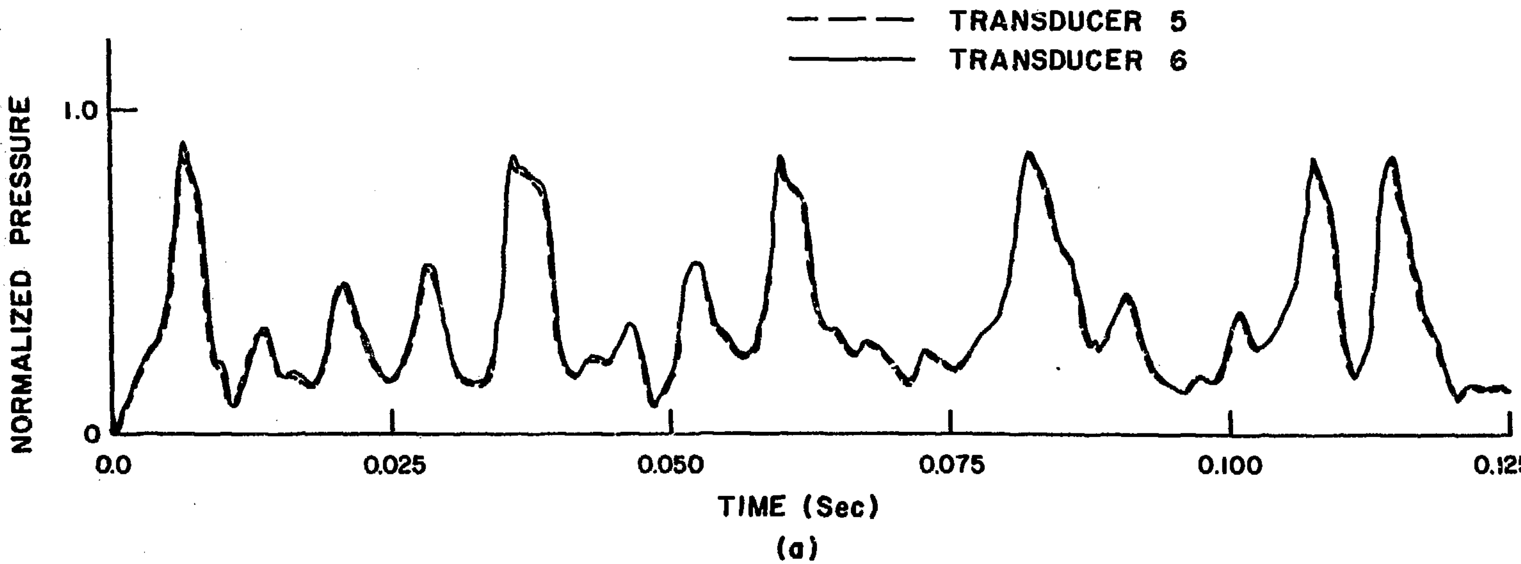
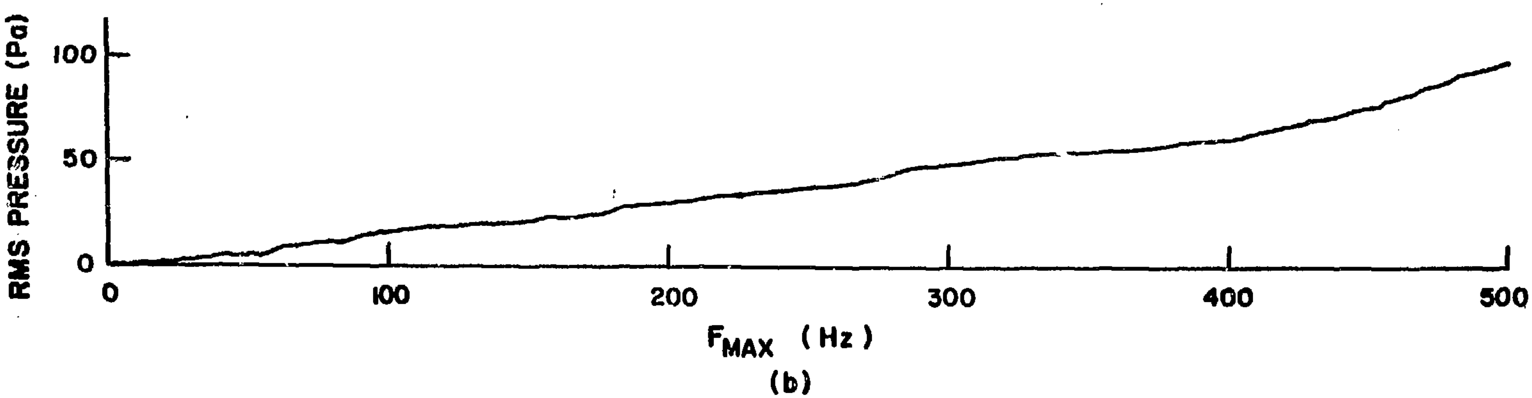

SIGNAL CONDITIONER
CH. A
FFT
ANALYZER
DATA
COPY
CH. B
TAPE
DECK
PLASTIC CHANNEL
PRESSURE TRANSDUCERS
TEST ROD
SPEAKER
NOISE
GENERATOR

Fig. 7

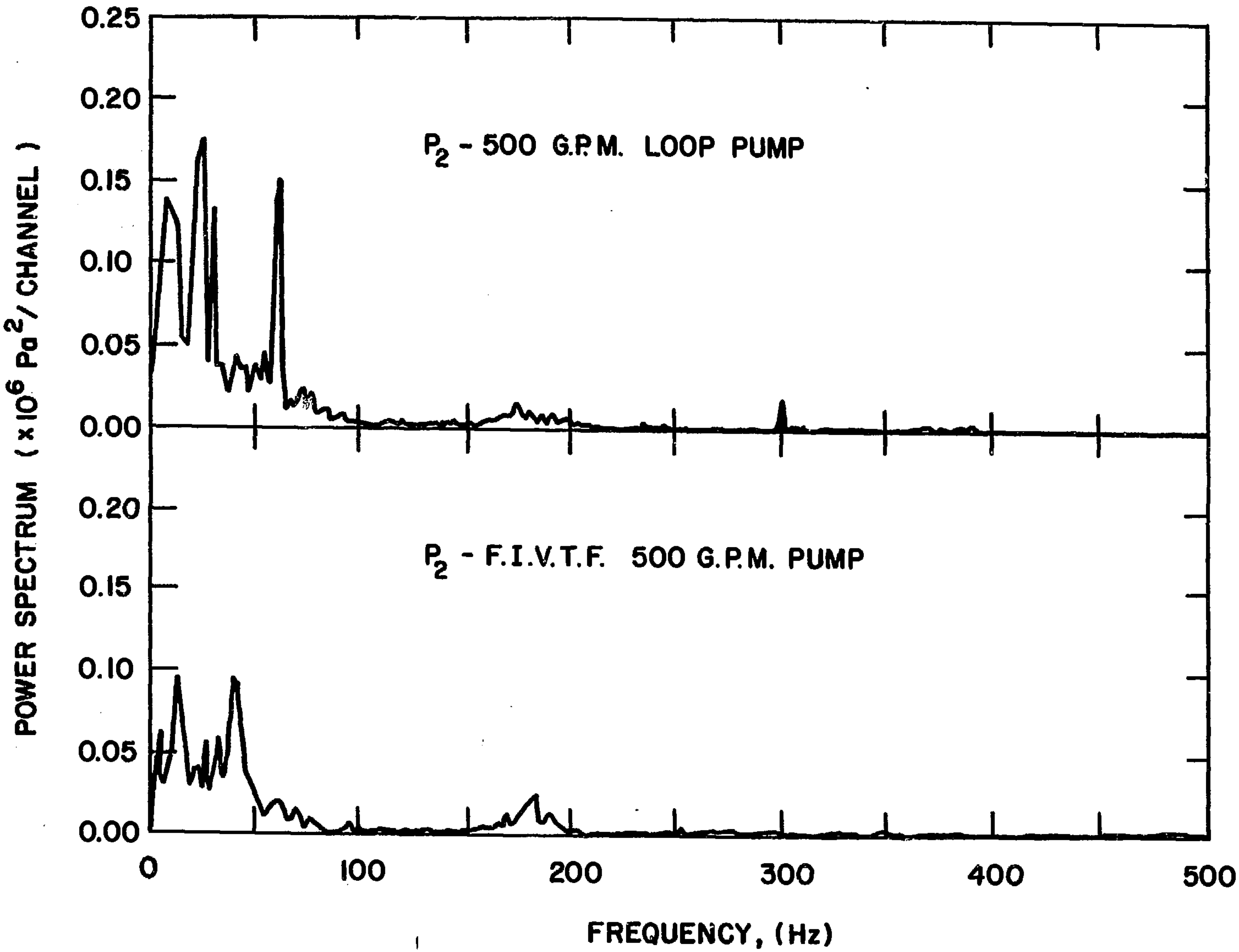


TRANSDUCER 5
TRANSDUCER 6
NORMALIZED PRESSURE
1.0
0
0.0
0.025
0.050
0.075
0.100
0.125
TIME (Sec)
(a)



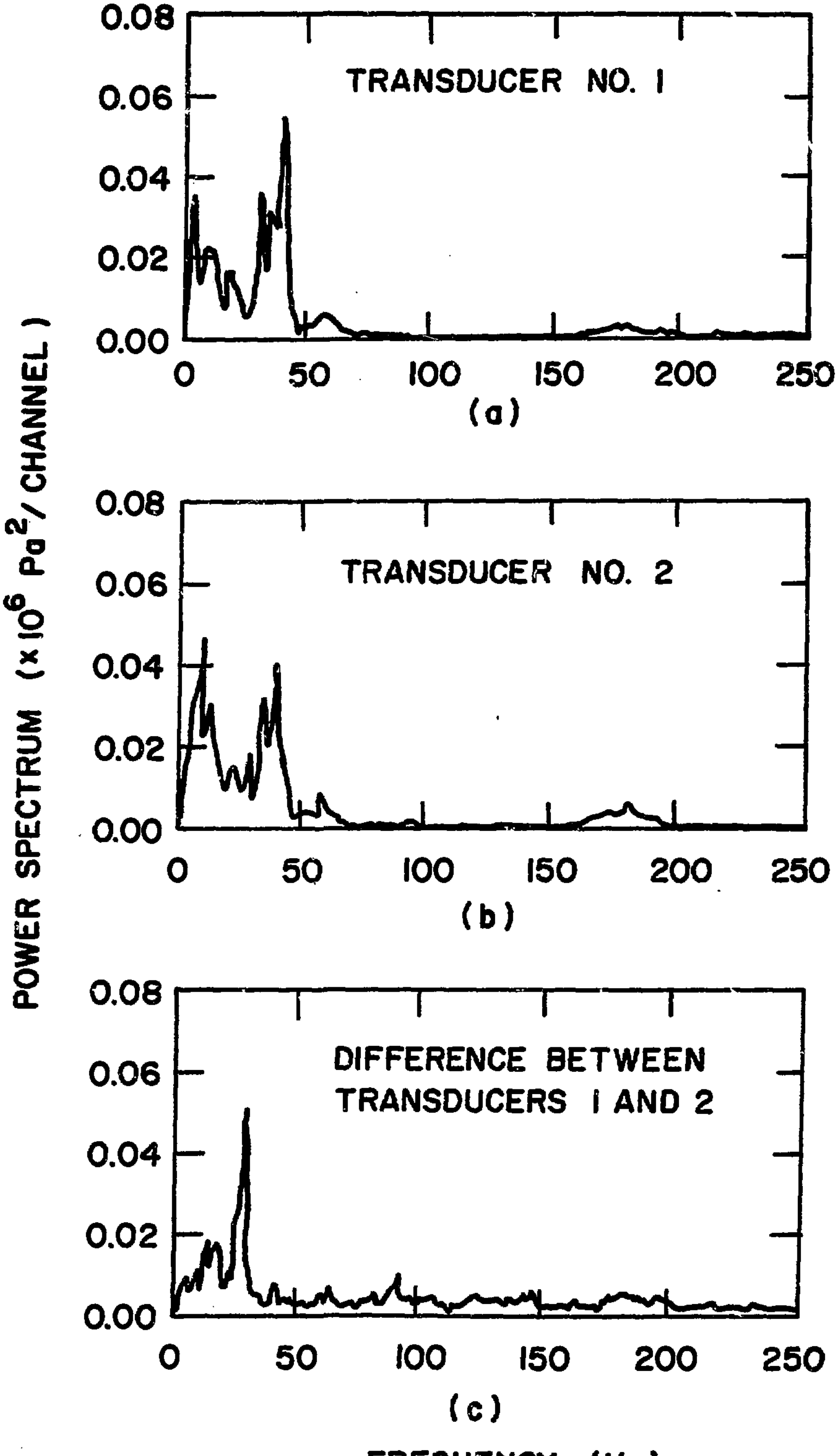
RMS PRESSURE (Pa)
100
50
0
0
100
200
300
400
500
F_{MAX} (Hz)
(b)

Fig. 8



POWER SPECTRUM (×10⁶ Pa²/ CHANNEL)
0.25
0.20
0.15
0.10
0.05
0.00
P2 - 500 G.P.M. LOOP PUMP
P2 - F.I.V.T.F. 500 G.P.M. PUMP
0
100
200
300
400
500
FREQUENCY, (Hz)

Fig. 9



POWER SPECTRUM (x10^6 Pa^2/ CHANNEL)
0.08
0.06
0.04
0.02
0.00
TRANSDUCER NO. 1
0
50
100
150
200
250
(a)
TRANSDUCER NO. 2
(b)
DIFFERENCE BETWEEN
TRANSDUCERS 1 AND 2
(c)
FREQUENCY, (Hz)

Fig. 10